ELECTRO-HYDRAULIC POWER STEERING SYSTEM

This is a regular United States Patent Application filed pursuant to 35 U.S.C. Section 111(a) and claiming the benefit under 35 U.S.C. Section 119(e)(1) of United States Provisional Application Serial No. 60/117,890, filed Jan. 29, 1999 pursuant to 35 U.S.C. Section 111(b)

FIELD OF THE INVENTION

This invention relates generally to power steering systems and more particularly to an electro-hydraulic vehicle power steering system incorporating an electric/hydrostatic steering assist module.

BACKGROUND AND SUMMARY OF THE INVENTION

Typical power assisted steering systems in use today include a belt-driven high rpm rotary hydraulic pump, specifically engineered hoses, tubes, couplings, and an array of brackets and fasteners and a rack and pinion subassembly. All of these components are engineered to endure the rigors of extreme thermal cycling brought about by a combination of ambient temperatures in the engine compartment, and various operational loads handled by the steering pump under the usual driving conditions.

Such power-assisted systems are a source of noise, operating inefficiency, and leakage, and consume a large amount engine power.

Power assisted steering pumps are built to very exact tolerances. Many components of these pumps are fabricated under tightly controlled manufacturing processes in order to maintain design specifications. Small discrepancies in manufacturing processes can lead to many performance problems.

One object of the system of this invention is to limit, if not entirely eliminate, many of the problems associated with present power steering systems.

Another object is to provide a power steering system which is composed of a relatively few simple parts, is rugged and durable in use, and is capable of being inexpensively manufactured and readily installed.

These and/or other objects, features and advantages of the invention will become more apparent as the following description proceeds, especially when considered with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a semi-diagrammatic side elevational view of an exemplary first embodiment-electro-hydraulic power assisted steering system constructed in accordance with the invention, showing a solenoid valve closed.

FIG. 2 is an enlargement of the solenoid valve within a circle in FIG. 1, showing the solenoid valve open.

FIG. 3 is a diagrammatic view of a controller for the system embodiment of FIGS. 1 and 2.

FIG. 4 is a semi-diagrammatic view of the hydraulic cylinder/actuator piston/motor screw drive module of the system of FIG. 1 as modified to incorporate a feed back encoder to serve as an actuator position sensor in an exemplary but preferred second embodiment system illustrated in FIGS. 4, 5 and 6.

FIG. 5 is a fragmentary duplicate of the left-hand portion of FIG. 1 as modified in the second embodiment system for incorporation of a steering wheel position sensor and a steering wheel torque sensor.

FIG. 6 is a diagrammatic view of the second embodiment of a power assisted steering system of the invention with a controller and associated bi-directionally data coupled multiplexed vehicle network.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

First Embodiment System

Referring now more particularly to the drawings, FIG. 1 shows, in an exemplary first system embodiment of the invention, an elongated rack **9** adapted to be connected at opposite ends to the steerable wheels (not shown) of a motor vehicle. The rack has a series of teeth **10** along a section of its length. A rotatable pinion **12** in mesh with the rack teeth **10** is connected to a steering wheel **13** of the vehicle by a shaft **14** so as to receive operator steering input.

The rack **9** extends lengthwise within an elongated housing **16** which is constructed to form an elongated power cylinder **18** adjacent one end. Spaced apart circular interior portions **20** and **22** of reduced diameter define the ends of the cylinder and have a sealed engagement with the rack in order to close opposite ends of the cylinder. A piston **24** carried by the rack within the cylinder **18** has a sealed engagement with the interior of the cylinder and separates the cylinder into cylinder chambers **26** and **28** on opposite sides of the piston. Hydraulic fluid fills the chambers **26** and **28**. A fluid line **30** communicates with the chamber **26** adjacent one end of the cylinder. A fluid line **32** communicates with the chamber **28** adjacent to the opposite end of the cylinder.

A hydraulic cylinder **40** is sealed at opposite ends by end caps **42** and **44**. A linear drive screw **46** extends lengthwise within the cylinder **40** and has its opposite ends rotatably received in the end caps **42** and **44** in a manner which permits the drive screw to be rotated in both directions about its central longitudinal axis, but preventing axial displacement thereof. The drive screw extends through and is threadedly engaged with an actuator piston **48**. The actuator piston is reciprocable within the cylinder **40** with its outer periphery in sealed engagement with the inner wall thereof, separating the cylinder into chambers **50** and **52**. Rotation of the drive screw **46** produces linear movement of the actuator piston in one direction or the other, depending on the direction of rotation of the drive screw. The drive screw extends through the end cap **44** and is externally connected to a suitable conventional servo motor **54** by a coupling **56**. The motor **54** may include a gear box and is carried by a housing **58** mounted on the end cap **44**. As an alternative, the end cap **44** may incorporate the motor and gear box.

The mating threads of the drive screw **46** and actuator piston **48** are engineered and fabricated to provide a smooth, low friction operation with minimal leakage between chambers **50** and **52**. However, a small controlled leakage of fluid between the drive screw and actuator piston is desirable because the fluid acts as a lubricant between the drive screw and the actuator piston. A subsequently generated boundary layer of fluid aids in limiting back lash typically associated with threaded components. Fluid leakage along the helical path is minimal. It is also predictable and of little consequence to a proportional control system.

The fluid lines **30** and **32** communicate with the hydraulic cylinder **40** through the end caps **42** and **44** leading into the cylinder chambers **50** and **52** on opposite sides of the piston **48**. Hydraulic fluid fills the chambers **50** and **52** and is fed

to and from the opposite sides of the rack piston **24** within cylinder **18** by motion of the actuator piston **48** under control of the motor **54** and drive screw **46**.

A fluid cross-over line **60** connects the fluid lines **30** and **32**. A solenoid valve **62** (FIG. **2**) is provided in the line **60**. The valve **62** has a valve element **64**, normally held open by the spring **66**, but closed by energization of the solenoid **68**. When the valve **62** is open, fluid is merely displaced from one side of the rack piston **24** to the other. Pressure sensors **70** and **72** are provided in the respective lines **30** and **32**.

In operation, the vehicle operator provides a steering input to rack **9** by means of pinion **12**. Motion of the rack and consequent motion of the piston **24** within cylinder **18** creates a pressure differential in chambers **26** and **28** which is sensed by the pressure sensors **70** and **72**. An electronic controller **74** (FIG. **3**), which may be controlled by the D.C. power system of the vehicle, receives the pressure signals and provides a control signal to motor **54** so as to command rotation of the drive screw **46** and consequent motion of the actuator piston **48** in a direction to pump hydraulic fluid from one of the chambers **50, 52** of the hydraulic cylinder **40** into one of the chambers **26, 28** of the power cylinder **18** to minimize the pressure differential between the two sensors. This electro-motivally developed motion of the actuator piston **48** and consequent hydraulically-developed fluid flow forces provide the power to assist the vehicle operator in manually applying torque via steering wheel **13** to achieve desired motion of the rack **9** to thereby move the steerable vehicle wheels.

When the vehicle is operated at a relatively low speed, it is essential that the power steering system be effective. However, at higher speeds, power assist is not demanded. Under such circumstances a vehicle speed sensor will input a signal to the electronic controller, whereupon the controller will de-energize the solenoid **68**, opening the valve **62** and disabling the power assistance. However, in an emergency situation, such as when the operator of the vehicle makes a sudden lane change, a momentary increase in fluid pressure in one of the chambers **26, 28** of the power cylinder **18** would be sensed by one of the sensors **70, 72** sending a signal to the controller to energize the solenoid **68** to close the valve **62** and allow the power assist to return to normal operation.

When initially starting a vehicle, it is desirable for the steering system to know the positions of the rack piston **24** and the actuator piston **48**. For example, the rack and wheels may have been turned to one extreme position after the vehicle was shut down. The actuator piston may, at this time, be centered. FIG. **1** illustrates a rack position encoder **80** including an elongated magnetic strip **82** connected to and extending lengthwise of the rack **9** and, a reader, such as a Hall sensor **84**, carried by the rack housing **16** for reading the rack position. A similar encoder, or alternatively, a rotary-type encoder **85/87** such as that illustrated in FIG. **4** and described hereinafter, may be provided to read the position of the actuator piston **48**. When starting the vehicle, the electronic controller **74** receives a signal from the sensor **84** of the encoder **80** for the rack piston and a signal from the sensor of the encoder for the actuator piston. The results are compared by the controller and if necessary the controller will activate the motor **54** and open valve **62** to rotate the lead screw **46** in the appropriate direction to move the actuator piston **48** into a position corresponding the position of the rack piston **24**.

The controller **74** may be powered by tie D.C. power system of the vehicle. The controller preferably consists of a micro-controller in the form of an Application Specific Integrated Circuit (ASIC). The ASIC preferably includes an integrated digital signal processor and appropriate analog-to-digital and digital-to-analog converters.

External inputs to the controller preferably are:

1. Vehicle Ignition Status (On/Off);
2. Pressure Sensor Inputs (Right/Left turn);
3. Rack Position Encoder;
4. Actuator Piston Position Encoder;
5. Vehicle Speed Sensor.

Internal inputs to the controller are preferably in the form of status bits wherein the system would have the ability to monitor its own health. This can be accomplished by comparing known real time output values to expected values found in look-up tables. Examples preferably are:

1. Resistance to impedance measurements on rotor windings;
2. Amperage required to achieve a particular torque value; and
3. Rotor winding temperature.

Digital output from the controller is preferably in the form of:

1. Digital voltage and amperage values for the linear actuator motor **54**. The sign of digital voltage value would indicate polarity and thus rotation position.
2. System status bits would be made available to the user by component monitoring purposes.

Digital values received from the controller are preferably converted to useable analog values by way of a analog-to-digital converter. The controller and linear actuator electronics are preferably optically isolated for overall circuit and system protection.

In a startup situation, the ignition is turned on and the controller **74** executes a startup procedure. The controller receives a signal to de-energize the solenoid **68**, opening the valve **62**, and also receives signals from the sensor **84** of the encoder **80** for the rack piston **24** and from the sensor of the encoder for the actuator piston **48**. The results are compared and the controller will then activate the motor **54** to rotate the lead screw **46** as needed to move the actuator piston **48** into alignment with the rack piston **24**, so that the positions of the two pistons correspond.

Upon alignment of the two pistons, the controller **74** energizes the solenoid **68** to close the valve **62**, so that the power assist is operative. The controller will execute a health status check. If all systems are verified and in proper working order, the controller Will report that the system is ready to receive steering input with power assist. The starting procedure may be completed in only a fraction of a second.

As a fail-safe feature the controller **74** will de-energize the solenoid, allowing spring **66** to open the bypass valve **62** in the event of a system failure, as, for example, a failure of the motor **54**, so that the vehicle operator will have complete control over the vehicle, but without power assist.

The actuator cylinder assembly can be manufactured as an individual component as described above for system modulation. Alternatively, similar system arrangement identical in operation, can be built as an integral part of the steering rack housing for system component integration.

Second Embodiment System

FIGS. **4, 5** and **6** illustrate, in conjunction with FIGS. **1** and **2**, an exemplary but preferred second embodiment of an electric/hydrostatic steering assist modular system of the

invention that retains the basic concept of the first embodiment steering assist module but adds extra capability with additional sensors and electronic communication of data over the multiplexed vehicle network in order to improve the performance, reliability and the ability of the system to adapt to future technology. Those components previously described in conjunction with the first embodiment system of FIGS. 1–3 are given like reference numerals and their description not repeated in referring to the second embodiment system.

The second embodiment system includes the pair of identical fluid pressure transducers **70** and **72** described previously (see FIG. 1) and utilized as described previously in the first embodiment system such that a pressure differential between the opposing hydraulic chambers **26** and **28** indicates quantitatively the input from the steering wheel **13** applied by the vehicle operator to rotate the pinion **12** and gear drive the rack **9** to the right or left to effect vehicle steering. It has also been determined in accordance with the invention that these sensors are also capable of sensing subtle input to the system by road-induced phenomena imparting forces back into the system through the steerable vehicle wheels and associated tie-rod ends.

Likewise the second embodiment system incorporates the absolute position encoder for the rack comprising the magnetic strip **82** and hall sensor **84** (FIG. 5) which again are utilized to determine the real time position of rack **9**. This data is used to determine the positional relationship of the rack to that of the steering wheel **13** and the actuator piston **48** in the steering system module **40–56** of FIG. **4** (see also FIG. **1**). Again, the real time position of the actual piston **48** is determined by an actuator piston absolute position encoder, which as indicated previously, may be of the type used on the rack, i.e., similar to the magnetic strip **82** and hall sensor **84**. Alternatively, FIG. **4** illustrates the incorporation of a conventional feed back encoder subassembly **85–87** made up of an emitter disk **85** affixed to drive screw **46** within end cap **44** for direct rotation with screw **46**. A sensor and emitter module **87** is stationarily affixed within end cap **44** and has a suitable gap for permitting direct rotary travel of a peripheral portion of disk **85** therethrough. The construction of emitter disk **85** and sensor and emitter module **87** is conventional and available commercially to provide accurate indicia of the angular displacement occurring between emitter disk **85** and linear displacement of actuator piston **48** in its range of travel in cylinder **40** in response to rotation of drive screw **46**. One example of a feed back encoder of this type is disclosed in U.S. Pat. No. 5,704,250 which in turn references U.S. Pat. No. 4,019,616, both of which are incorporated herein by reference and therefore not further described. The second embodiment system incorporates additional sensors as diagrammatically illustrated in FIG. **5**, namely a conventional steering column absolute position encoder **90** that senses angular rotation of steering wheel **13** as inputted to steering column **14**, and a conventional steering column torque sensor **94**.

The steering wheel absolute position encoder **90** provides the following steering wheel information:

a. The angular displacement in degrees (left or right) from the center position. The center position is defined as the point where the steerable wheels are straight ahead.

b. The rate at which the steering wheel is being turned (measured in degrees per second).

The steering wheel position information from sensor **90** is preferably analyzed in the second embodiment system and used for:

a. Initialization and positioning of actuator piston **48** in the steering assist module at the time of vehicle start-up.

b. All steering maneuvers.

The steering wheel position information is thus used to calculate the required rpm of the electric motor **54** for steering assist operations.

During vehicle operation a measurable amount of torque is applied to steering column **14**, either proactively by the vehicle operator through steering wheel **13** and/or reactively to road forces reflected back through the steering gear of the vehicle into the system. This torque value reflected in steering column **14** is affected by a number of factors including:

a. The coefficient of friction between the vehicle tires and the driving or road surface. This coefficient of friction in turn is affected by:
   1. Vehicle weight,
   2. Vehicle speed, and
   3. Driving surface conditions (i.e., dry pavement, surface temperature, gravel, sand, water, ice).

b. Friction between components of the mechanical steering system:
   1. Articulating joints (i.e., steering column universals, bearings, tie-rod ends, ball joints);
   2. Mating gear surfaces;
   3. Lubrication and contamination seals.

c. Continued application of force to steering wheel after:
   1. Design travel limits of steering system have been met (i.e., fall turn left or right);
   2. Contact with an external obstruction (e.i., curb or a rut in the driving surface).

d. Continued application of steering force to offset external forces;
   1. Constant radius turns (i.e., ramp onto freeway);
   2. Driving surfaces that pitched perpendicular to direction of travel (i.e., a crowned road).

In the second embodiment system the data obtained through torque sensor **94** is used in conjunction with data taken from the original pressure transducers **70** and **72** and integrated to determine and control the magnitude of the torque output of motor **54** to be applied to the develop the hydraulic fluid pressure to assist vehicle steering operations. This data can also be used to differentiate between operator input and road induced phenomena through the suitable software systems of controller **74**.

As indicated diagrammatically in the system layout of FIG. **6**, the previously described electronic controller **74** can be bi-directionally coupled by an exchange patch connection **100** to a conventional multiplexed vehicle network **102**. The signals from pressure sensors **70** and **72** are inputted to controller **74**, as are the signals from rack position sensor **87** and actuator piston position sensor **84**. The output from controller **74** again is developed to control both motor **54** and solenoid valve **62**. However, as seen in FIG. **6**, the on/off input signal of the system is applied to the multiplexed vehicle network **102**, as is the vehicle speed data, instead of directly to controller **74** as in the first embodiment system per FIG. **3**. In addition, the steering torque data from sensor **94** as well as the steering wheel position data from sensor **90** are inputted initially to the multiplexed vehicle network **102**. There then is the bi-directional exchange of system data over the multiplexed vehicle network via patch link **100**. Thus the multiplexed vehicle network will share with the conventional vehicle on-board ECU the power steering system status, pressure sensitive data, rack position data and actua-

tor sensor data. Accordingly, system performance can be monitored by a suitable software, system diagnostics can be analyzed by a suitable software, and system performance can enhanced by changes in such software.

As indicated hereinbefore, it will be understood that, in both the first and second embodiments of the electric/hydrostatic steering assist system of the invention, the system designer has the option of integrating the power assist module **40–54** with the rack housing **16**, such as by piggy-back mounting thereon or encapsulated by suitable re-design of housing **16**. When the system components are thus integrated with the rack housing the system can be shipped in a charged state, i.e., filled with steering fluid, so as to eliminate fluid handling processes at the vehicle assembly plant, in contrast to conventional power steering systems that require such costly final assembly fluid charging operations.

On the other hand, either of the first and second system embodiments can be separated into the main sub-assemblies consisting of the power module **40–54**, the solenoid valve **62**, and the rack and pinion steering gear components **9–28**, and **80**, **84** of FIG. **1**. When modularized into separate sub-assemblies the hydraulic electromotive unit module can be conveniently mounted in a suitable remote locations on the vehicle, and likewise the solenoid valve **62** and associated sensors **70** and **72** can be conveniently located along the fluid coupling lines **30** and **32** as best suits the vehicle application installation. This flexibility of the system of the invention thus reduces the vehicle packaging constraints as compared to that of conventional hydraulic power steering units as well as electric steering technologies currently available.

From the foregoing description it will now be apparent that the electic/hydrostatic steering assist system embodiments of the invention amply fulfill the aforestated objects and provide many features and advantages over the prior art. As will be apparent to those of ordinary skill in the art, the system embodiments provide an alternative replacement for conventional rotary hydraulic pump driven steering technologies. The embodiment components of the system reduce costs by eliminating the high tolerance machine components required in such rotary pump driven systems, reduces the part count required in the components for a power steering system, and reduces potential leak points compared to conventional steering systems. In addition, the system provides an improvement in average fuel economy, provides a more versatile alternative selection to the pure electric steering systems currently available while providing all the functionality of such pure electric steering systems through the computer software in vehicle electronic controller units as well vehicle multiplexed networks. The system of the invention also retains steering system compliance, i.e., the parameter defined as the customary "feel" of the steering system. In addition, the steering assist system is harmlessly disabled via the fail-safe spring loaded spindle valve **64** of the controller valve unit **62**.

As indicated previously, the system performance can be monitored via suitable software in the integration with the controller and the multiplexed vehicle network. The system diagnostics can likewise be analyzed via such software, and the performance of the system can be readily enhanced via changes in such software.

The development of hydraulic force for assisting in steering by use of the hydraulic ram type power unit **40, 46, 48**, with mechanical force multiplication obtained via the pitch of lead screw **46** as well as the option of a gear reduction unit coupled between the output shaft of motor **54** and lead screw **46**, enables high hydraulic pressure development in the hydraulic system while avoiding the noise, vibration and harshness (NVH) factors commonly encountered with conventional steering systems. In such systems equipped with the usual high rpm rotary hydraulic pump, the pressure pulsations developed by the pump output have been a constant problem in terms of undesirable system noise generation and transmission in the hydraulic lines, often with first, second and even third order harmonics of pump output frequency being related and amplified by hydraulic transmission system resonant frequencies. All of these problems are eliminated by the present system.

The system of the invention is also adaptable to a variety of land, sea and air vehicles, and can be applied to both manned vehicles as well as remotely and autonomously controlled vehicles. As indicated in the second embodiment system of FIGS. **4**, **5** and **6**, the acquired data from the sensors of the steering system can be utilized by other systems on the vehicle via the multiplexed vehicle network **102**, e.g., traction control, anti-lock braking, active suspension and collision avoidance vehicle systems.

Motor **54** can be of the type set forth in the aforementioned U.S. Pat. No. 5,704,250, namely an economical conventional frameless motor with permanent magnet rotors to provide a high torque-to-weight inertia ratio with built-in hall effect devices utilized for electronic commutation. Such motors can be of the sequence pulse/rotary stepping servo type so that accurate control over the degree of rotation is produced when the windings are properly pulsed. In this manner, the system is effective to move the fluid from one or the other of the chambers **50** and **52** of the hydraulic cylinder **40** by accurately controlled linear travel of the actuator piston **40** therein. The force by which this fluid is moved is directly proportional to the difference in pressure between chambers **26** and **28** as sensed by the sensors **70** and **72**.

Thus, as indicated previously, in low speed or parking conditions, the torque to the system is greatly increased as the pressure differential between chambers **26** and **28** is likewise increased as well. As the vehicle speed increases, the steering load decreases. The load to the system may decrease to the point where power assist is no longer necessary. Under these conditions the solenoid actuated spindle valve **64** is system opened when the speed of the vehicle reaches a point where power assist is no longer required. Fluid can then flow from chamber **26** to chamber **28**, and vice versa via line **60** and the by-pass valve **64**. Since valve **64** is spring biased toward open position, if the power assist device fails or electrical power is lost to the solenoid, the biasing spring of valve **64** drives spindle **64** to the open condition so that the vehicle operator then can have complete manual control over the vehicle until it can be brought to a safe stop.

In the event of an emergency lane change where a large input to the system is created by the vehicle operator, a momentary pressure increase would be sensed by the pressure transducers **70** and **72**, thereby resulting in the closing of the by-pass valve **62** to thereby assist the driver with steering control until the vehicle driving situation is returned to normal.

The system computer software and associated position sensors of rack **9** and actuator piston **48** enable the rack and actuator piston to be adjusted so that the fluid volumes on either side of the actuator piston **48**, i.e., in chambers **50** and **52**, are proportional to the fluid volumes in rack assist chambers **26** and **28** on the respective sides of the rack piston **24**. This enables the linear actuator piston **48** to be properly